IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Appl. No. : 10/761,865 Confirmation No.:3149

Applicant : Koneda et al. Filed : January 21, 2004

T.C./A.U. : 3748

Examiner: Chang, Ching Docket No.: 81044248 Customer No.: 33066

DECLARATION UNDER 37 CFR 1.131

Mail Stop Amendment T.C./A.U. 3748 Commissioner for Patents P.O. Box 1450 Alexandria, VA 22313-1450

We, James Ervin, Thomas Megli and Philip Koneda, state that:

- 1. We are joint inventors of the above-identified patent application;
- We made an invention claimed in the above-identified patent application in the United States prior to February 4, 2003.
 - 3. In support thereof, we submit the following evidence:

A. A Document (see attached) created by the above-identified James Ervin entitled "DISCLOSURE OF VALVE ACTUATOR USING A HYDRAULIC DISPLACEMENT AMPLIFIER" by James D. Ervin, Thomas Megli and Philip Koneda, the date on said Document having been removed but such date being prior to February 4, 2003, said Document being attached as attached EXHIBIT A;

B. Said Document shows on page 3 in Figure 3 a hydraulic lever similar to FIG. 3 of the above-identified patent application;

C. On or about January 01, 2003, it was decide to initially fabricate the valve shown in Figure 3 of Exhibit A with the two pistons forming portions of the walls of Application No.: 10/761,865

the chamber having the equal surface areas (i.e., a 1:1 ratio device). More particularly, we chose to make a 1:1 ratio device as a matter of convenience because it allowed us to utilize our pre-existing hardware and supporting test fixtures. Although this prototype was realized at 1:1, ALL of our modeling and analysis documentation was focused on characterizing behavior on ratios greater than 1:1. For our purposes, the real value of hardware is to validate the relationships being modeled and the 1:1 hardware was convenient for that purpose (confirm leakage / tolerance / viscous relationships, etc.). We then rely on the models to allow us to predict behavior at the ratio condition that is most appropriate for a given application, package space, etc.

- D. A Document dated February 25, 2003 attached as EXHIBIT B, which are slides presented at a design review of the hydraulic lever described in the document in Exhibit A; designed at ratio 1:1. This hardware was fabricated between February 25th and July of 2003,
- E. Exhibit C is a print out of a screen shot of a computer directory of the above-identified James Ervin showing that on March 17, 2003 a document was created entitled "patent review for hydraulic level and vibration cancellation.ppt";
- F. Exhibit D is a copy of the document referred to in paragraph C above and entitled "patent review for hydraulic level and vibration cancellation.ppt";
- G. Exhibit E is a copy of a shot of the above-identified Thomas Megli's computer screen showing analysis of the hydraulic lever shown on the second pages of the document in Exhibits D such shot such seen being shown dated in the upper right hand corner April 25, 2003.
- H. Between April 25, 2003 and July 24, 2003 we focused on engineering assessment of various prospective suppliers. This work was essential as Ford relies on supplier partnerships to commercialize technology such as the hydraulic lever. In this search, the supplier is engaged at an exploratory level to assess whether the proper complement of skill, experience, and manufacturing capability are complementary with our long term need. The nature of these assessments can be gleaned though the file titles from this period from Exhibit "O": e.g. Actuator collision3.xls evaluating the role of impacts between mechanical components, 6 bar open from M1.dat evaluating the challenge of

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actuating under gas forces, LSP JCAE comparison slide.ppt – internal high level system supplier comparison, supplier selection technical tradeoffs.ppt – internal technical system supplier comparison, TRW armature failure analysis translation.doc – internal component supplier evaluation, LSP power consumption estimate 072403 5.xls – projecting power consumption for comparative selection.

- I. Exhibit F shows on the first page thereof a computer screen shot indicating that on July 24, 2003 computer aided design (CAD) was refined for the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.
- J. Exhibit G is a copy of a page from a notebook of the aboveidentified Thomas Megli dated July 28, 2003 showing calculations related to the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.
- K. Exhibit H is a copy of a page from a notebook of the aboveidentified Thomas Megli dated July 29, 2003 showing oil feed flow rate data measured for the prototype shown in exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.
- I.. Exhibit I is a copy of a page from a notebook of the aboveidentified Thomas Megli dated July 30, 2003 showing calculations related to the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas.
- M. During the period between February 25, 2003 and November 11, 2003, activity was towards building, testing, and refining a prototype of the valve shown in the last 3 sheets of Exhibit E, such valve being the valve with the two pistons forming portions of the walls of the chamber having the equal surface areas. Typically the design process begins with a concept sketch that is reviewed with the CAD designer. The designer then converts the concept sketch into a realizable CAD cross-section that delivers the functions desired. This process of creation and review can take several weeks, and must consider how best to the implement the critical features that will be demonstrated with a hardware confirmation prototype. For the hydraulic lever, the critical feature to be

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demonstrated was the ability to transmit force, thru the hydraulic path and return the armature to the fully released position. This was best implemented by converting existing magnetic circuit hardware from earlier prototype hardware. The design creation process, converting from sketch to realizable hardware typically takes about I month. After that the design is further refined to include secondary features, for the hydraulic lever this included tailoring the ports to ensure the oil transfer between the two chambers was stable. Once the design model was complete, we sent the model to a local machine shop who then built all of the details, from the model. This process took about 1 month. Once the hardware was received, we had to wait for an opportunity to test in our laboratory. This could have taken 1-2 months. Remember that our "internal concept development" was a secondary activity to our primary responsibility of identifying a suitable first tier supplier for this project. During assembly and debugging in July of 2003, we discovered that since the parts were made directly from the model, critical component dimensions, like the clearance between the hydraulic pistons and the housing was excessive and internal leakage was too high for adequate function. The piston dimensions were revised a new pistons fabricated adding another 1-2 weeks. Again, some delay to find time to test, and finally the testing that verified the feasibility of this concept. All in all, this additional time could have taken 2 to 4 months.

- N. Exhibit J is a copy of a Management Performance Review reporting work on the hydraulic lever concept, including general modeling for ratios > 1:1 and testing of the 1:1 ratio device identified above. This was presented to our management on November 11, 2003 by the above-identified Thomas Meeli with portions being removed.
- O. Exhibit K is a copy of a Management Performance Review reporting work on the hydraulic lever concept, including general modeling and reporting for ratios > 1:1. This was presented to our management on November 11, 2003 by the aboveidentified James Ervin with portions being removed.
- P. Exhibit L is a copy of a screen print of a listing of files having data taken on the above-identified 1:1 ratio device by the above-identified Thomas Megli dated July 2 and July 3 2003.
- Q. Exhibit M is a screen shot of the test data taken on the aboveidentified 1:1 ratio device taken on July 2 and July 3, 2003. Note that "Date Created" is later

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in time because the files were copied from one computer to another. "Date Modified" dates show the last time modifications were made to the file.

- S. Exhibit N are screen print outs showing creation of various files generated directly support the engineering development of the hydraulic actuator, various systems integration work (such as electrical system analysis, wiring analysis, magnetic material property research), and the EVA (electromagnetic valve actuation) supplier selection process (which was a critical step for commercializing EVA) described in Figure 3 of Exhibit A. Such files compiled for James Ervin's work as example activity, and being dated between the period of time between February 6, 2003 and January 24, 2004.
- 4. All statements made herein of my own knowledge are true and that all statements made on information and belief are believed to be true.
- 5.1 understand and have been advised that willful false statements and the like made herein are punishable by fine or imprisonment, or both (18 U.S.C. 1001) and may jeopardize the validity of the application or any patent issuing thereon.

Date: 7/19/06

James Erv

Date: 7/19/06

5

DISCLOSURE OF VALVE ACTUATOR USING A HYDRAULIC DISPLACEMENT AMPLIFIER

James D. Ervin, Thomas Megli, Philip Koneda

Introduction

One common approach to controlled valve actuation is to use two electromagnets to toggle an armature connected to a valve between an open or closed position where it is held, while a pair of springs is used to force the valve to move (oscillate) to the other state (Fig. 1). A common shortfall to such a design is that the magnets must generate force over a travel length equal to the valve stroke. At points of travel where the air gap is large, more current is required to achieve a given force resulting in a corresponding increase in power consumption. Conversely, the peak force that can be generated for a given current is reduced as the air gap increases, which effectively reduces the authority to control the valve motion.

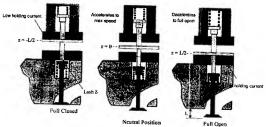


Figure 1: Conventional linear direct acting spring-mass oscillator
In contrast to the direct acting mass-oscillator approach, a new actuation architecture
is proposed here which utilizes a hydraulic lever amplify the motion of a magnetic

armature to achieve a desired valve displacement, thus reducing the effective air gap. This disclosure describes the potential claims regarding this new architecture:

- · Use of a hydraulic lever to minimize variation in air gap.
- Incorporation of passive hydraulic lash adjustment in the hydraulic lever mechanism.
- Incorporation of passive hydraulic damping in the hydraulic lever mechanism to potentially enable open loop control.
- Incorporation of a bypass valve to tune the damper response and allow for initialization of the system at startup.
- Ability to flexibly package the system to meet package requirements.

Prior art

As a departure from the direct acting systems, there is at least one example of a lever actuated system: LSP innovative Automotive Systems of Munich, Germany has developed an actuator that uses a mechanical lever to amplify the travel of the armature and reduce the effective air gap. In contrast to the hydraulic system proposed here, the mechanical lever does not address issues of passive lash management or passive damping and must be designed within specific packaging rules. As a result of the reduced gap, the LSP lever system does improve the control authority through the stroke and improves the power consumption relative to conventional linear oscillators.

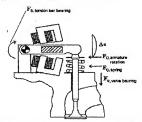


Figure 2: LSP lever acting spring-mass oscillator

Overview of the hydraulic amplifier lever oscillator

As the subject of this disclosure, a hydraulic lever is proposed as an alternative means of amplifying armature displacement. This has the effect of reducing the armature travel that is required to achieve a desired valve displacement and, in turn, reduces the effective air gap. In one proposed arrangement (Fig. 3), a first hydraulic piston is attached to an armature and biased with a first spring to be normally held in a downward position while a second piston is attached to a valve and biased with a second spring in a normally upward position.

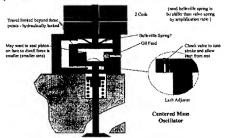


Figure 3: Hydraulic lever acting spring-mass oscillator

During a startup sequence, an upper coil is used to pull the armature upward. This creates higher pressure in the outer cavity than the central cavity to open a check valve mounted on the second piston, allow fluid transfer from the outer cavity to the central cavity, and compress the first spring.

Following this initialization process, the upper coil is de-energized, causing the first spring to move the armature and first piston downward. This increases the pressure in the central cavity between the piston faces, decreases the pressure in the outer cavity, and closes the check valve. The pressure difference across the second piston causes it to move downward and compress the second spring. At some time during this process, the lower coil is energized to continue compressing the second spring until the second piston strokes beyond the point where it can communicate oil with the outer cavity. At this

time, the second piston becomes hydraulically locked, travel stops, and the valve is held in the open position. Alternately, the travel limits could be regulated by designing the first piston to strike a shoulder in the central cavity or by driving the armature to land on the lower coil. are expected to be that is controlled by opposing coils.

Conversely, the lower coil can be de-energized and the upper coil can be energized to reverse the process and close the valve. It is expected that the prossure in the central cavity will be greater than that of the outer cavity during this event so that the check valve remains closed. This expectation requires that the force from the upper coil minus the force from the upper spring be less than the force of the second spring at all times and can be ensured by designing the preload of the second spring appropriately.

Lash Adjustment

The use of hydraulic lash adjusters for controlling tolerance stackup and thermal growth in valvetrains is well established. Each cycle following the scating of the valve, a controlled leak (orifice) from an oil lube reservoir is used to charge a hydraulic chamber to eliminate the clearance between the valve stem and the cam profile. In the case of EVA, this lash adjuster would be positioned between the valve stem and the actuation source. When acted upon by the cam, the oil becomes trapped in the cavity (can't leak out fast enough) and pushes on the valve with essentially no lash. The key to providing lash adjustment in this manner is designing the controlled leak to be slow enough to allow only a minimal change in length during the valve stroking process while being fast enough to account for the rate of length change due to thermal growth. It must also operate across a wide range of viscosity as the oil temperature.

Referencing the hydraulic lever implementation shown in Fig. 3, hydraulic lash control could be readily achieved by designing the appropriate clearance between the support body and the upper and lower pistons. In a typical sequence of operation, the lower piston would stroke upwards until the valve became seated. The upper piston would then continue stroking until its travel became limited by the seating of the armature or by a closing off of the channel that communicated oil to the underside of the lower piston (hydraulic locking). During this event, a differential pressure would develop across the upper piston, causing fluid to flow into the central cavity through the check valve and leak between the perimeter of the upper piston and the support body to resolve

the volume displaced. Having charged the cavity, the actuation source would be able to begin opening the valve with no lash.

During an opening event, downward motion of the upper piston would cause the lower piston to stroke downward, compressing the valve spring and creating a differential pressure across the lower piston. As a consequence of the differential pressure, fluid would flow out of the central cavity, resulting in a lower net stroke of the lower piston than for the upper piston. Such lost stroke is actually desired to account for valve growth due to thermal effects, where the loss of stroke (leakage) is ideally designed to be greater than the maximum thermal growth that can occur during a given cycle.

As a tradeoff during the valve closing event, the length lost during opening would result in the valve landing before the upper piston had finished stroking. With the natural coupling of position and velocity for the upper and lower pistons, it is advantageous to design the leakage to be as small as possible so that the travel of the two pistons is nearly the same. Thus if one is landed gracefully, both will, leading to simpler control.

Passive Damping

Taking advantage of the hydraulic architecture, it is also simple to incorporate passive damping into the actuator. As an example (Fig. 4), the travel of the pistons could be damped at the travel extremes by having the pistons close off a port, stopping the communication of oil to the rest of the circuit. At this point the system would be hydraulically locked though a conservation of volume. A check valve could be incorporated into the overtravel portion of the cavity to facilitate the release event. In a further refinement, the shape of the port could be tailored to provide a desired level of damping as a function of piston travel (ref. docket 201-1552 Variable Area Damper), as suggested in Fig. 4. Other implementations could include a ring extending from the piston and engaging mating cavity on the support (ram damper).

Passive, velocity-dependent damping offers significant advantages over active EVA control:

 Reduces or climinates the need for high speed, complex position and current feedback control of the EVA solenoids — This complex control is presently required to achieve soft valve seating velocity for present actuator systems. The feedback control requires a high-speed (approximately 10-20kHz control loop frequencies) computer, a position sensor for every engine valve/EMVA assembly. The control algorithms required for soft-landing are highly nonlinear, and require complex structures, such as adaptive or iterative learning control schemes to compensate for changes in actuator and valve characteristics over the life of the engine.

2. Improves system robustness and repeatability – The damper reacts to remove nearly all of the armature kinetic energy near the end of a valve transition; therefore, the damper achieves low contact velocities for a wide range of solenoid voltage control inputs. This robustness demonstrates that the damper will compensate for changes in operating characteristics due to manufacturing variability, engine wear, fluctuations in vehicle supply voltage, and changes in gas flow force disturbances.

As a drawback, such passive damping does dissipate energy and will increase power consumption relative to the undamped case. It should be noted however that active control methods to manage impact events, such as lash take-up and landing, increase power consumption as well. The magnitudes of these effects are still under investigation.

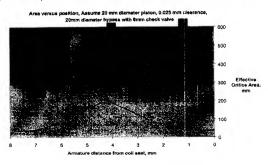


Figure 4: Example Implementation of Passive Damper

Vibration Cancellation

For mass-oscillating EVA systems, the typical valve acceleration and combined valve/armature mass is greater than for conventional cam designs leading to increased engine vibration. In an effort to minimize this effect, a design variant of the hydraulic lever is proposed where the upper port of the upper piston communicates with the upper port of the lower piston and the lower ports are similarly connected. This arrangement forces the motion of the two pistons in opposite directions, so that the inertial loads of each tend to cancel according resulting in a net force:

$$F_{net} = m_1 a_1 - m_2 a_2$$

where m_1 and m_2 are the masses of the upper and lower pistons and a_1 and a_2 are their respective accelerations.

Considering an example using armature and valve masses from conventional linear oscillator technology, a peak force of 630N would be produced by a 400g accel acting on the 0.090Kg upper mass (armature / stem / spring) and 0.073Kg lower mass (valve / keeper / retainers / spring). If the masses were driven in opposition by a hydraulic amplification, the net force would instead be 66N, a tenfold reduction. If a larger amplification were considered, experience with the LSP lever system suggests that the armature mass would need to increase in keeping with the lever ratio while the associated acceleration would decrease.

Donut and hole opposed

Variable orifice damping

Find armature (upper) mass as a function of design parameters and valve (lower) mass:

$$\Delta u_{post} = \pi \sqrt{\frac{L \times SF \times TR}{M_v}} \frac{M_v}{M_v} + \frac{1}{TR^2}$$

$$\frac{M_v}{M_u} = \frac{2 \times FPAM \times (\Delta u_{post})^2}{L \times SF \times \pi^2 \times TR} \frac{1}{TR^2} = \frac{2 \times FPAM \times (\Delta u_{post})^2 \times TR \times sign(TR) - L \times SF \times \pi^2}{L \times SF \times \pi^2 \times TR^2}$$

$$M = \frac{M}{TR} \frac{\{L \times SF \times \pi^2 \times TR^2\}}{\{L \times SF \times \pi^2 \times TR^2\}} \frac{1}{TR^2} \frac{1}{TR^2}$$

Significant mass ratio relationships:
$$\frac{M_{\phi}}{M_{\psi}} = \frac{\left(L \times SF \times \pi^{2} \times TR^{2}\right)}{\left(2 \times FPAM \times \left(\Delta I_{mass}\right)^{2} \times TR \times sign(TR)\right) - \left(L \times SF \times \pi^{2}\right)} = \frac{A}{B - C}$$

 Lowest TR that can be applied for a given transition time (TR_{min}): Armature mass goes to when TR satisfies the relationship:

$$B-C=0 \Rightarrow TR \times sign(TR) = \frac{L \times SF \times \pi^2}{2 \times FPAM \times (\Delta t_{max})^2}$$

Armature mass also goes to cowhen TR goes to co

 Find TR which produces the minimum armature mass (TR_{opt}): Armature mass is minimum when:

$$\frac{\partial M_{S}}{\partial (TR)} = 0 = M_{\gamma} \left(\frac{2(L \times SF \times \pi^{2} \times TR)}{(L \times PPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times 2ign(TR))(L \times SF \times \pi^{2} \times TR^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})} \cdot \frac{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}{(2 \times FPAM \times (\Delta_{Spec})^{2} \times TR \times sign(TR) - L \times SF \times \pi^{2})}$$

$$\frac{\delta M_{d}}{\delta(TR)} = 0 = M_{\pi} \left\{ \frac{\left(2 \times L \times SF \times R^{2} \times TR \frac{M}{2} \times FPAM \times (\Delta_{mage})^{2} \times TR \times ign(TR) - L \times SF \times R^{2}\right) \cdot \left\{FPAM \times (\Delta_{mage})^{2} \times ign(TR) \times TR^{\frac{M}{2}}\right\}}{\left(2 \times FPAM \times (\Delta_{mage})^{2} \times TR \times ign(TR) - L \times SF \times R^{2}\right)^{2}} \right\}$$

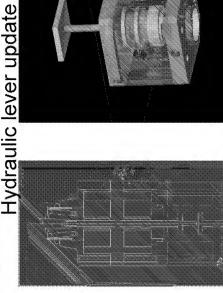
$$\frac{\delta M_{d}}{\delta(TR)} = 0 \Rightarrow TR \times sign(TR) = \frac{L \times SF \times R^{2}}{FPAM \times (\Delta G)} \cdot \frac{1}{V}$$

Notice that TRoot is exactly twice the value of TRmin-

- As TR is reduced from the TR for min armto the lowest applicable TR value, the effective mass is growing like
- 4) As TR approaches cofrom the TR for minimum armature mass, the armature mass increases because more actuator force is required to deliver the same force to the valve.

Actuator and Power Electronics Design and Engineering Review

Actuator Concept Development





ATDL flexible test rig for hydraulic lever with integrated damping



February 25, 2003

Actuator Concept Development

Conclusions

- One mass damper results confirm earlier results documented contact velocities < 0.1 m/s
- Smaller diameter piston had better than expected smaller damper to further reduce mass and midimpact velocity performance → may allow even travel losses
- Some unexplained temperature effects



Actuator Concept Development

Next Steps

- Complete testing and analysis of 1 and 2 mass dampers (ongoing with other ATDL testing)
- prove out lash adjustment and damping functions Build and test hydraulic lever prototype rig to (4-03)
- Initiate detailed package study for optimized system on AJ133 cylinder head(3-03)



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Name	520	Type	Modified	Created
203-0185 patent		File Folcier	7/31/2003 1:	7/17/2003 9:55 AM
203-0185 base lever patent		File Folcier	9/19/2003 11	7/31/2003 1:17 PM
*** SEVA Actuator with Hydrosife Displacement Amplifier.ppt	60 KB	Microso t	4/10/2002 1:	4/3/2002 3:29 PM
EVA Actuator with Hydraulic Displacement Amplifier2.ppt	225 KB	Mcrosa t	12/16/2002 5	6/24/2002 12:40 PM
Description of the Hydraulic Lever Invention.doc	20 KB	Mcroso't	7/22/2002 11	7/22/2002 11:57 AM
Description of the Hydreuk: Lever Invention2.doc	1,271 KB	Maroso't	8/6/2002 2:1	7/22/2002 12:28 PM
Description of the Hydraulic Lever Invention3.doc	1,302 KB	Maroso't	10/3/2002 9:	8/6/2002 2:16 PM
Description of the Hydraulic Lever Invention 4.doc	161 KB	Marosoft	11/7/2002 6:	10/3/2002 10:58 AM
It briefs auto Lever Invention amplification lash take up demong for patent doc	1,259 KB	Microsoft	10/29/2002 1	10/28/2002 10:59 AM
Method to cancel vibrations in EVA actuators doc	117KB	Microsoft	12/16/2002 2	10/29/2002 10:42 AM
Hydrausic Lever Invention for controlling two valves doc	1,259 KB	Microsoft	10/30/2002 5	10/30/2002 9:36 AM
Method to optimize lever for min trens time.doc	134 KB	Microsoft	12/16/2002 3	11/7/2002 6:03 PM
# 188135 hyd lever lesh take up demping disclosure.doc	1,267 KB	Microsoft	11/10/2002 e	11/18/2002 8:32 AM
Hydraulic lever EVA description for 88136 121002.doc	1,274 KB	Microsoft	12/16/2002 1	12/10/2002 1:57 PM
Driving multiple valves from one actuator 121602 doc	78 KB	Microsoft	1/2/2003 4:0	12/16/200Z 2:39 PM
- patent review for hydraulic lever and vibration cancellation.ppt	1,400 KB	Microsoft	3/17/2003 12	3/17/2003 12:03 PM
# 19764(MethodocencelvibrationsinEVAactuetors2.doc	154 KB	Microsoft	6/25/2003 4:	6/25/2003 3:49 PM
TAPGTI-080PLS-scewed Roures 1-5.pdf	257 KB	Adobs Ac	7/7/2003 11:	7/7/2003 11:35 AM
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1976+0MethodtocencelvbrationsinEVAactuetors3.doc	151 KB	Microsoft	7/9/2003 4:3	7/9/2003 4:32 PM

How the "Hydraulic Lever Actuator" overcomes the shortcomings of Conventional "Direct-Acting Oscillator" Actuators:

Conventionally need electromagnets to act over large working gap

-> large inductive change = hard to control

/ lever reduces working gap

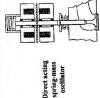
Transition time is limited by fundamental laws of force / armature mass and stiffness

iotos / almaure mass and sumioss
-> engine performance is limited
-/ lever reduces effective mass – faster transition

Closed loop control of "soft landing" is difficult -> questionable robustness / reliability / incorporate damping into hydraulic circuit to improve control of landings

Mechanical lash requires use of energy consuming soft-release strategy -> inefficient

✓ incorporate hydraulic lash take-up

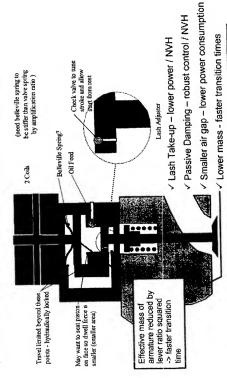






A Proposed Implementation of the Hydraulic Lever Oscillator

Hydraulic lever acting spring-mass oscillator





Effect of Lever on Magnetic Efficiency

Direct acting has lower consumption during hold since lever is a force divisor, but holding consumption is less than 10% of total consumption

Lever reduces working gap to reduce current required to generate desired force.

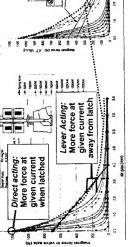
Current reduces by a factor of the lever ratio for modest to large gaps to reduce consumption during transitions.

Net power savings can be substantial – more than 50% for lever ratio > 2.

LSP Lever Actuator

Direct Acting Oscillator

(Hydraulic Lever would have the same effect)





Passive Damping

Passive Damping Implementation Example:

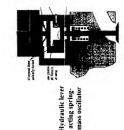


"opposed masses" in a Hydraulic Lever Actuator Vibration cancellation in EVA actuators through

- Conventional EVA actuators generate large shaking forces due to relatively high mass and fast transition time
- -> much higher shaking forces than non-EVA valvetrain – especially at idle -> can't reduce transition time without
 - unctional loss

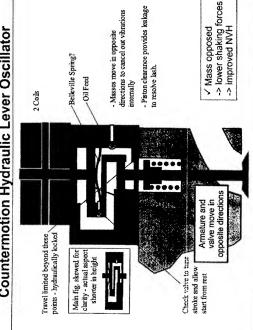
 can't reduce mass due to physical
 initiations
- Hydraulic lever can be configured to allow armature and valve to move in opposite directions to produce net force reduction / elimination

Direct acting spring-mass oscillator





Countermotion Hydraulic Lever Oscillator A Proposed Implementation of the





Quantitative Vibration Force Improvement

0000 8 8 8 8 8 Reduce forcing from 700N (direct Example For 2.9ms Transition: acting) to 0N (-2 lever ratio)! F, HM, X Fra = Fy - Fa TR=- 12/15 Fz=Ma X / TR F.=M. X/TR £ 3

imature and valve mor in same direction ---2.7ms 2.9ms Example Net Vibration Force as a Function of Transmission Ratio Transmission Ratio and Transition Time Net Vibration Force as a Function of - Exhaust Valve Mass = 0.053Kg -99 200 300 -200 200 400



- 3.2ms -3.5ms

> Fransmission Ratio * xvalvaltam φ

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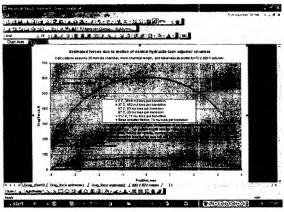
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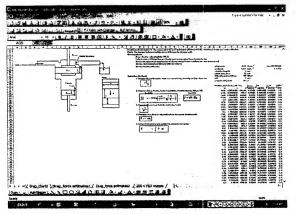
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TR = 1214

Third (lower) Class levers



Design analysis worksheet

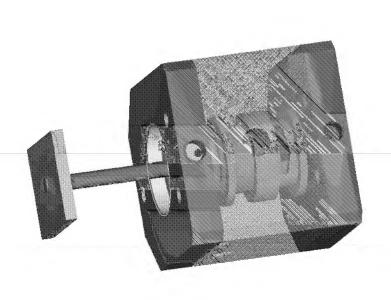


Design analysis worksheet

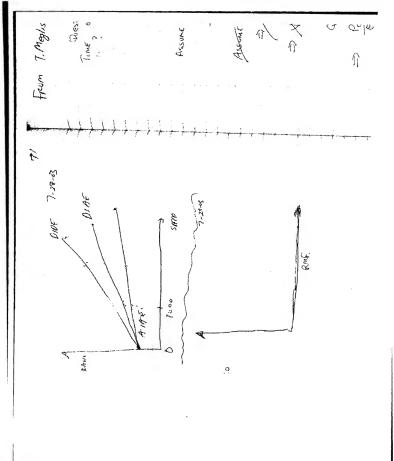
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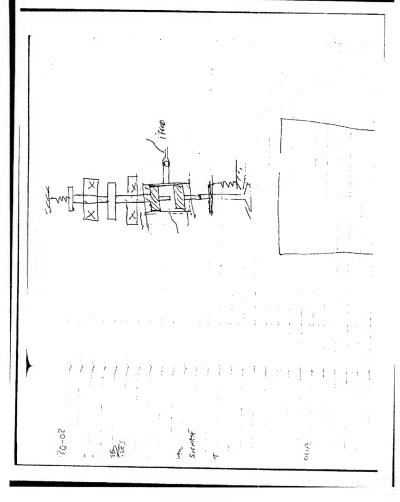


EXHIBIT J (3 Sheets)

Technical Specialist

Tord Meter Company, Management Performance Review

Name: Thomas W. Megli

Position:

National/Global ID:	Performance Period: 2003
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For further comments, go to the additional comments page attached

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ome: Thomas W. Megli	Position ⁻	Technical Specialist
ational/Global ID:		e Period: 2002
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EY RUSINESS OBJECTIVES	CONTRIB	UTIONS TOWARD OBJECTIVES Assessment
New actuator concept development (40%). Develop alternative actuator concepts with emphasis or hydraulic damper design, (1) revise and calibrate model and evaluate redesign to deliver function with minimal losses, (2) deliver revised design specifications and concepts to hardware design and build starm. (3) Support hardware testing and evaluation (O4). Other (20%): A. Support Dymota camiess wall-wetting model development (Q3), B. NVH torsional analysis and induction noise support	implemente model deve new actual to predict g function of rejection at lest data. New actual A Develop damper to impact on application November (contin	nued on additional comments sheet)
For further comments, go to the additional comments (maken with the same	***************************************
ASSESSMENT OF LEADERSHIP BEHAVIORS	(Based on leads	rack on Ford leadership behaviors)
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Additional Comments

KEY BUSINESS OBJECTIVES

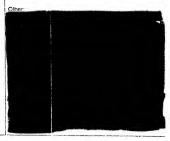
CONTRIBUTIONS TOWARD OBJECTIVES

New actuator concept development (conjugation

 Developed hydraulic damen bypass control concept that schedules landing velocity as a function of engine speed and oil temperature. Validated with hardware and and submitted invention disclosure.

D. Developed concepts for sitiograting hydrautic damping function into hydrautic fever actuator concept. 2 invention disclosures submitted and hydrautic lever design initiated.

 r. developed actuator dynamics model to support hydraulic lever actuator design



ASSESSMENT OF LEADERSHIP BEHAVIORS (Bused on feedback on Ford saudurship behaviors)

ıme: James D. E	rvin	Position:	LL6 - Technical Expert	
ational/Global ID:		Performan	ce Period: 2003	
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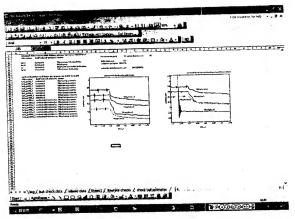
Additional Comments

Contribute to development of internal actuation concept to prove function and quality write reducing cost (Q2). Lipport development of a production oriented system (Q4). Develop Ford IP and submit disclosures. (Q4) > Prepared patent applications for hydraulic lever a vibration cancellation and disclosures for minimum I power and valve mask with mini lift. VCR disclosure submitted. TCVT petent issued. ASSESSMENT OF LEADERSHIP BEHAVIORS (Bessed on feedback on Ford leadership behaviors)	BJECTIVES CONT	RIBUTIONS TOWARD OBJECTIVES
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power and valve mask with mini lift. VCR disclosure submitted, TCVT petent issued.	melity while reducing cost (Q2). including	ared documentation of hydraulic lever concept ag disclosures and patent applications (Q2-3).
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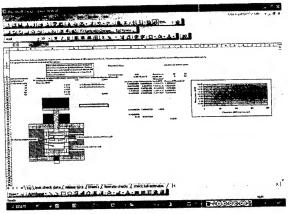
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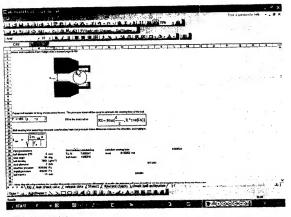
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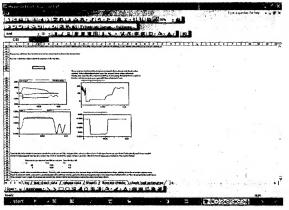
Another worksheet with test data



More data and analysis on leakage flow



Check ball design analysis



Dymola modeling results of prototype actuator with lever.

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